NATIONAL BUREAU OF STANDARDS REPORT

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EFFECT OF VARIATION OF EMISSIVITY OF INTERNAL SURFACES OF HEATED BOX BEAMS ON TEMPERATURE DISTRIBUTION, THERMAL STRESS AND DEFLECTION

by

S. Goodman, S. B. Russell and C. E. Noble

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EFFECT OF VARIATION OF EMISSIVITY OF INTERNAL SURFACES OF HEATED BOX BEAMS ON TEMPERATURE DISTRIBUTION, THERMAL STRESS AND DEFLECTION[1]

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1. ABSTRACT

Transient temperature distribution histories, thermal stresses, and deflections were computed for 13 box beams uniformly heated along one cover. Various heating rates, geometries of beam cross-section, and thermal properties were considered. Heat transfer was by radiation and conduction. Gaseous heat transfer and possible effects of yielding, creep, and buckling were neglected.

For maximum beam temperatures above 700 - 900° F, change in emissivity of the interior surfaces of the beam had an appreciable effect on the cover (but not the web) temperatures and, to an even greater extent, on the beam deflection. At maximum beam temperatures of 1200° F, an increase in interior surface emissivity caused an appreciable decrease of the maximum thermal stress.

A rough experimental check of temperature distribution and beam deflection was made for one case.

2. INTRODUCTION

The subjection of aircraft and other structures to increasingly high temperatures gives rise to a need for predicting transient temperature distribution in the structures, and the resulting thermal stress and deformations. This knowledge will make possible the rational design of the structure, and the devising of methods of alleviating of thermal stresses and deflections. With increasing structure temperature,

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[1] This work was conducted at the National Bureau of Standards under the sponsorship and with the financial assistance of the Office of Naval Research.

radiation increases in importance as a mode of heat conduction[1][2]. The influence of radiant heat transfer on the temperature distribution of the structure is affected by structural geometry, heating rate, and thermal properties of the beam material. It was the purpose of this investigation to determine the effect of variation of emissivity of the interior surfaces of a box beam on temperature distribution, maximum thermal stress, and beam deflection in heated box beams for a variety of conditions.

3. STRUCTURES AND MATERIAL

The structures considered were three box beams 30 in. long--two relatively thick-walled and one relatively thin-walled. The beam cross-section is shown in the figure at the top of table 1; wall thicknesses are indicated by the ratios listed in columns 2 and 3 of the table. One beam (cases 1, 2 and 3) is a thick-walled beam, similar to a beam used in the experimental work. The other beams are typical of beams used in aircraft structures; one relatively thick-walled (cases 4-8) and the other relatively thin-walled (cases 9-13).

Thermal and elastic properties were taken as those of the type 302 stainless steel test beam. Thermal conductivity and specific heat were approximated by the linear relationships:

$$k = 7.08 + 0.0043T \qquad (1)$$

$$c = 0.106 + 0.0000257T \qquad (2)$$

where

k is thermal conductivity (Btu/hr ft °F)

c is specific heat (Btu/lb °F)

T is absolute temperature (°R)

Emissivity was taken as 0, 0.35 (emissivity of the test beam, cases 1-3) and 1, as indicated in columns 4 and 5 of table 1.

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[2] Figures in brackets indicate the literature references at the end of this paper.

Young's modulus and the coefficient of linear thermal expansion were represented by

$$E = 31.0 \times 10^{6} - 0.0064T \times 10^{6} \quad (3)$$

$$\alpha = 8.29 \times 10^{-6} + 0.00137T \times 10^{-6} \quad (4)$$

where

E is Young's modulus (lb/in.^{2})

α is coefficient of linear thermal expansion (in/in. °F)

T is absolute temperature (°R)

4. BOUNDARY CONDITIONS

The beams were considered to be subjected to a uniform heat input along one cover in a rarefied atmosphere. The heating rates are listed in columns 6 and 7 of table 1. The initial beam temperature was 78° F. For the test beam (cases 1, 2, 3), the boundary condition was taken as the heated cover temperature (figures 2 and 7). For all other cases, the boundary condition was taken as the heat input to the cover. All elements of the heated cover were considered to remain constant at 1200° F after reaching that temperature.

5. METHOD OF ANALYSIS

5.1 Calculation of Temperature Distribution

Analysis was made by a numerical method similar to that described by Dusinberre (ref. 2). One half of the cross-section of the symmetrical beam was divided into 12 analysis elements. Since the external heat transfer was symmetrical with respect to the left and right halves of the beam cross-section, the right half of the beam was represented by a reflector. The reflector was divided into four regions. The element configuration for cases 1, 2 and 3, shown in fig. 1, is typical of all cases.

A heat balance equation was set up for each analysis element and solved for the element temperature after a short time interval, $\Delta\theta$. The process was repeated for successive short time intervals using the

new element temperature as the starting point of the next computation. It was assumed that surface emissivity remained constant and that beam surfaces were gray, i.e., thermal reflection is diffuse and emissivity and absorptivity are equal even when the temperatures of the incident radiation and of the receiver are different. The heat balance equation is as follows:

$$\begin{aligned} T_m^1 = T_m + \frac{\Delta\theta}{V_m \rho C_m} \Big\{ & (k_{m-1,\, m})(H_m)(T_{m-1} - T_m) \\ & + (k_{m,\, m+1})(J_m)(T_{m+1} - T_m) + \epsilon_m \sigma A_m (T_A^4 - T_m^4) \\ & + \sigma A^1_m \left[F_{m,\, 1} (T_1^4 - T_m^4) + F_{m,\, 2} (T_2^4 - T_m^4) + \dots \right. \\ & \left. \dots + F_{m,\, 12} (T_{12}^4 - T_m^4) \right] + W_m \Big\} \end{aligned} \qquad (5)$$

where

T_m^1 is the temperature at the center of element m after time interval $\Delta\theta$ (°R)

V_m is the volume of element m (ft^3)

T_m is the initial temperature at the center of element m (°R)

ρ is the density of the material (lb/ft^3)

C_m is the specific heat of element m (Btu/lb °F), represented by a linear function of T_m.

$k_{m,\, m+1}$, $k_{m-1,\, m}$ are the averaged thermal conductance of element m and adjacent element (Btu/hr ft °F)

H_m, J_m are ratios of element contact areas to distances between element centers for element m and adjacent elements m - 1, m + 1 respectively (ft).

ϵ_m is emissivity of element surface m

σ is the Stefan-Boltzmann constant
$= 1.713 \times 10^{-9}$ (Btu/ft^2 hr °R^4)

A_m is exterior surface area of element m (ft^2)

T_A is ambient temperature (°R)

A^1_m is interior surface area of element m (ft^2)

$F_{m,n}$ is an overall radiant heat interchange factor for net radiant heat exchange between a surface of element m and a surface of element n. It includes the effect of direct and all reflected radiation.

W_m is the external heat input (Btu/hr).

The first two terms inside the brackets on the right hand side of eq. 5 represent conductive heat transfer between element m and adjacent elements; the third term represents radiant heat transfer with the outside environment, the fourth term represents internal radiant heat exchange between element m and the rest of the beam interior, the fifth term represents the boundary condition of a (variable) heat input. For cases 1, 2 and 3, in which the boundary condition was taken as the heated cover temperature history, the temperature of each heated cover element was expressed as a series of four linear functions of time covering successive time intervals.

To compute the gray body radiant heat exchange factors, F, first black body radiant heat exchange factors were computed for all two element combinations of the 16 internal surfaces of the analysis elements and (fictitious) reflector using the methods of reference 3, and radiation geometry. Values of the 256 F's were then obtained from the matrix relationships of reference 4. The computations were performed on SEAC using an existing code for inverting the matrices.

The temperature-distribution computations were computed on SEAC. Temperatures at the mid-points of the 12 analysis elements were printed out at regular time intervals which were whole number multiples of $\Delta\theta$.

In order to minimize computing machine running time, it was desirable to use the largest value of the time interval, $\Delta\theta$, consistent with adequate accuracy of solution. The time interval must be sufficiently short that during the interval (1) the initial element temperatures can be used with negligible error and (2) for conductive heat transfer, the effect of any non-adjacent element on a given element is negligible. Satisfactory values of $\Delta\theta$ were obtained by trial. A portion of the temperature

distribution history was computed several times using successively larger values of $\triangle\theta$. The largest value of $\triangle\theta$ which gave a temperature distribution history negligibly different from that obtained when the smallest value of $\triangle\theta$ tried was used to compute the complete temperature distribution history. Values of $\triangle\theta$ found satisfactory by this method ranged from 0.72 second for rapidly heated beams to 1.8 seconds for slowly heated beams.

Some error was generated in the beam temperature history computations by representing one-half of the beam cross-section by a gray, rather than a specular reflector. As a check on the magnitude of this error, radiant heat transfer rates were computed for a simple symmetrical case using first the entire beam cross-section in the computations, and then an equivalent beam consisting of one-half of the beam cross-section and a gray reflector. An infinitely long box beam of rectangular cross-section divided into six analysis elements was used. The beam was two inches deep, eight inches wide, and emissivity was 0.35. The vertical (two inch) walls of the beam were at temperature T_1 and the horizontal walls at absolute zero. The rate of radiant heat transfer to the left half of the upper horizontal wall was

$$\begin{aligned} q &= 0.1354\sigma T_1^4 \quad \text{for original beam} \\ q &= 0.1389\sigma T_1^4 \quad \text{for equivalent beam with reflector} \end{aligned} \tag{6}$$

where

q is rate of radiant heat transfer (Btu/hr ft^2)

σ is the Stefan-Boltzmann constant
(0.1713 x 10^{-8} Btu/(ft^2 hr °R^4))

T_1 is vertical wall temperature (°R).

It was concluded that the error due to use of a gray rather than a specular reflector was small.

5.2 Computation of Thermal Stress and Deflection

Thermal stresses and deflections in the beam were computed by a method similar to that described in reference 5. Integrations over the

beam cross-section were performed numerically using finite elements identical with those used in the temperature distribution computations (fig. 1).

6. TEST OF BEAM

The test specimen, corresponding to case 2, table 1, was a box beam constructed of 0.128 inch 18-8 type 302 stainless steel sheet (figure 1). Two pieces of sheet were each bent into identical channel shapes and butt welded together lengthwise to form the 2 in. x 5 in. x 30 in. beam.

The beam was heated in a vacuum chamber whose pressure was maintained at 4mm of mercury, a pressure low enough to eliminate convective heat transfer. Only one side of the beam was subjected to heating. It was heated with twelve quartz tube, tungsten filament heating elements uniformly spaced in a reflector. Total output of the heaters was 5.7 Btu/second. The reflector was a rectangular box enclosing the heating elements and fitted to the heated wall of the beam. It was constructed of stainless steel and silver plated to reflect a maximum of the heat output to the beam. Reflectivity of the reflector surface was about 0.97.

Temperatures were measured at the center cross-section on the heated cover of the beam, at the center cross-section on the unheated cover, at two points on the heated cover one in. from one end, and at one point on the heated cover one in. from the other end. The temperatures were measured with iron-constantan thermocouples whose outputs were indicated by galvanometers. The thermocouples were calibrated before and after the test. Temperatures of the heated cover at the beam ends were approximately 12 percent lower than the temperatures of the heated cover at the center cross-section. The experimentally determined heated cover temperatures, used as the boundary condition in case 2, are shown in fig. 2 and, together with observed temperatures for two points on the opposite cover, in fig. 7.

The total normal emissivity of the 18-8 (type 302) stainless steel was determined by comparing its rate of radiation at a given temperature with that of a Globar at the same temperature. The radiation rates were determined by focusing the images of equal areas of the Globar and of the stainless steel successively on a thermopile, using a flourite lens. The emissivity of the stainless steel obtained was constant in the temperature range 400° F to 1000° F and equal to 0.35.

Beam deflection at the center cross-section was measured by means of SR-4 type AB-5 electrical strain gages mounted back-to-back on a shielded cantilever beam which was deflected by displacement of the center

of the beam relative to its ends. Contact of the center of the beam with the cantilever was made by a Vycor rod. Calibration was accomplished by deflecting the cantilever by known amounts at the point of contact with a micrometer screw.

The maximum deflection observed was 0.415 in. after 240 seconds of heating. After 378 seconds of heating the center deflection was 0.37 in.

7. RESULTS

7.1 Beam Temperature Distribution

Computed temperature histories at point, A, in the center of the heated cover and at point, B, in the center of the opposite cover (see sketch on table 1) are shown in figures 2 to 6 for the 13 cases considered. Fig. 7 shows the computed temperature distribution in a thick-walled beam (cases 1, 2, 3) of the same dimensions as the test beam after 148, 189 and 297 seconds of heating. Fig. 8 shows the temperature distribution in a thin-walled beam after 45 seconds of slow heating, and after 13.7 seconds of fast heating. Values of A and B are listed in columns 9 and 10 of table 1. Comparison was made of geometrically similar beams, heated at the same rate for the same length of time but having the following different values of interior or exterior emissivities: 0, 0.35 and 1.0.

It was found that, in all cases considered, change in emissivity of the surfaces had an appreciable effect on the cover temperatures and comparatively little on the web temperatures when the maximum heated cover temperature was above 700 to 900° F. For beams of the same geometry and heating boundary condition and with heated cover at a temperature of about 1200° F, the temperature difference, d, between points A and B, is from 18 percent to 48 percent greater for cases having zero internal emissivity than for the corresponding cases having black body interior surfaces. Greatest difference is for a thin-walled beam with one cover heated at about 30° F/second; least difference is for a thin-walled beam with one cover heated at about 91° F/second.

A change in exterior surface emissivity had less effect on d than a change in interior surface emissivity. Comparison of cases 7 and 8 with corresponding cases 4 and 6 indicates that, for the particular conditions specified in table 1, increasing the interior surface emissivity from 0 to 1.0 causes a decrease in d of 19 percent; increasing the exterior surface emissivity from 0 to 1.0 causes a decrease in d of only 8 percent.

7.2 Thermal Stress and Deflection

In order to evaluate the effect of a change in emissivity on the thermal stress distribution and deflection, thermal stress distributions and deflections were computed for the 13 cases under the conditions listed in table 1. Maximum tensile stress, compressive stress and deflection are listed in columns 11, 12 and 13 respectively in the table. The possible effects of yielding, creep and buckling were neglected. Values of the thermal stress and deflection are therefore valid only for comparison purposes. It was found that for a thick-walled, slowly-heated beam with heated cover at 958° F (cases 1, 2, 3), a change in the interior wall emissivity had little effect on the magnitude of the maximum thermal stress. An increase in the interior wall emissivity from 0 to 1.0 however, resulted in a decrease of 34 percent in the maximum beam deflection. For the beams with the same geometry and the same heating conditions and in which the heated cover temperature was about 1200° F, a change of the interior wall emissivity from 0 to 1.0 reduced the absolute maximum stress from 9 percent (cases 4 and 6) to 16 percent (cases 9 and 11, 12 and 13), and deflection from 16 percent (cases 4 and 6) to 52 percent (cases 9 and 11). In the latter beam, reduction in maximum tensile stress was 36 percent. For a change of the exterior wall emissivity from 0 to 1.0 (cases 7 and 8), reduction in absolute maximum stress was 8 percent, and reduction in maximum deflection was 7 percent.

Generally, a change in the thermal gradient produced by a change in the interior beam emissivity from 0.35 to 1 is about 75 percent of the thermal gradient due to a change in the interior beam emissivity from 0 to 1. Corresponding ratios for maximum thermal stress and deflection are about 68 and 75 percent, respectively.

7.3 Comparison of Experimental and Theoretical Results

A partial check of the temperature distribution and deflection was made for case 2. Computed and observed temperatures for two points on the unheated cover after 148, 189 and 297 seconds of heating are shown in fig. 7. Agreement is fair. Some of the discrepancy may be attributed to gas heat conduction within the beam. For the conditions listed under case 2 in table 1, observed beam deflection was 0.37 in. and beam deflection computed from the theoretical temperature distribution was 0.38 in.

8. CONCLUSIONS

A change in the interior surface emissivity of a box beam heated along one cover has an appreciable effect on the temperature distribution,

and consequently on the maximum thermal stress and the deflection. For a variety of beam geometries and heating conditions, a change of the interior surface emissivity from 0.35 to 1 for a stainless steel beam reduces considerably the temperature gradient in the temperature range above 700° - 900° F and the maximum thermal stress and the deflections in the temperature range above 900° - 1200° F. The effect is particularly marked for a thin-walled slowly-heated beam.

For the Director,

B L Wilson
B. L. Wilson, Chief,
Engineering Mechanics Section,
Division of Mechanics.

Washington, D. C.

June 1958

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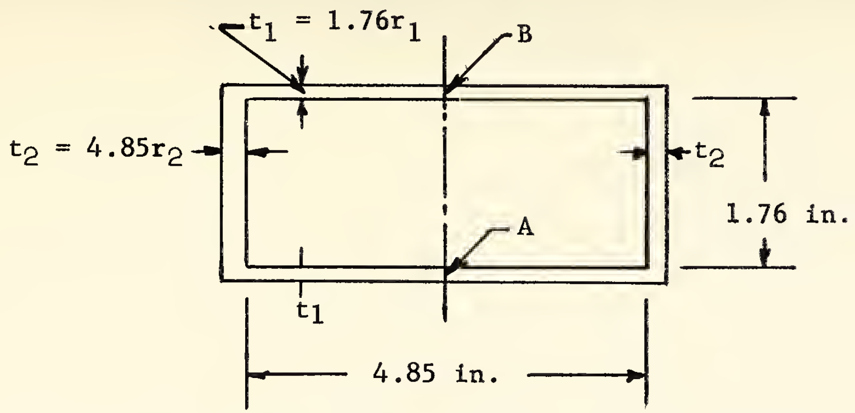
1. S. Goodman, Radiant-Heat Transfer Between Nongray Parallel Plates, J. Research NBS Vol. 58, No. 1, January 1957.
2. G. M. Dusinberri, Numerical Methods for Transient Heat Flow, Trans. ASME Vol. 67, No. 8, November 1945.
3. D. C. Hamilton and W. R. Morgan, Radiant Interchange Configuration Factors, NACA TN 2836, December 1952.
4. W. H. McAdams, Heat Transmission, Third Edition, McGraw-Hill, New York, 1954.
5. S. Levy, Thermal Stresses and Deformations in Beams, Aeronautical Engineering Review, Vol. 15, No. 10, October 1956.

Table 1 - Cover Temperatures, Maximum Thermal Stress and Deflection in Box Beams 30 Inches Long-Heated Along One Cover.



(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
			Surface emissivity		Heating rate			Temperature[2]		Maximum stress[3]		Maximum deflection[3], (in.)
Case	r_1	r_2	interior	exterior	(Btu/hr ft^2)	(°F/sec)[1]	Elapsed time (sec)	at A (°F)	at B (°F)	tensile (psi)	compressive (psi)	
										x 10^3	x 10^3	
1	0.0727	0.0264	0	0.35	-	3.8	378	958	84	35.3	29.3	0.41
2	"	"	0.35	"	-	"	"	"	147	35.8	29.7	.38
3	"	"	1	"	-	"	"	"	377	34.8	30.5	.27
4	0.02	0.03	0	"	32400	50	26.6	1202	78	121.7	127.9	.44
5	"	"	.35	"	"	"	"	1173	101	118.0	124.4	.42
6	"	"	1	"	"	"	"	1095	188	108.5	116.1	.37
7	"	"	.35	0	"	"	"	1200	102	121.5	127.1	.43
8	"	"	.35	1	"	"	"	1104	98	111.0	116.7	.40
9	0.01	0.01	0	.35	10800	30	45	1204	78	102.9	99.0	.52
10	"	"	.35	"	"	"	"	1135	154	92.1	91.0	.46
11	"	"	1	"	"	"	"	959	371	71.5	74.9	.31
12	"	"	0	"	32400	91	13.7	1200	78	127.6	101.1	.52
13	"	"	1	"	"	"	"	1125	203	114.5	95.1	.44

[1] approximate temperature rise rate at point A up to about 800° F

[2] initial temperature 78° F

[3] neglecting yielding, creep, and buckling

Notes:

Mat'l: Stainless steel, 18-8 (302)
Length: 30"
Analysis elements designated by dotted lines and small numerals.

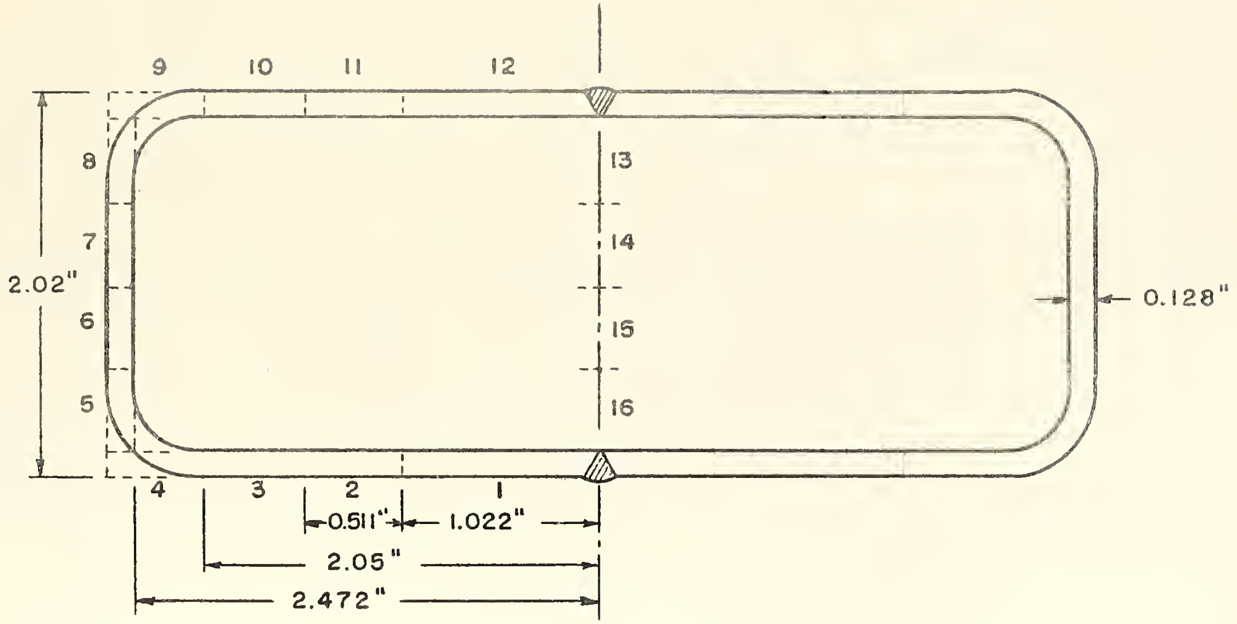
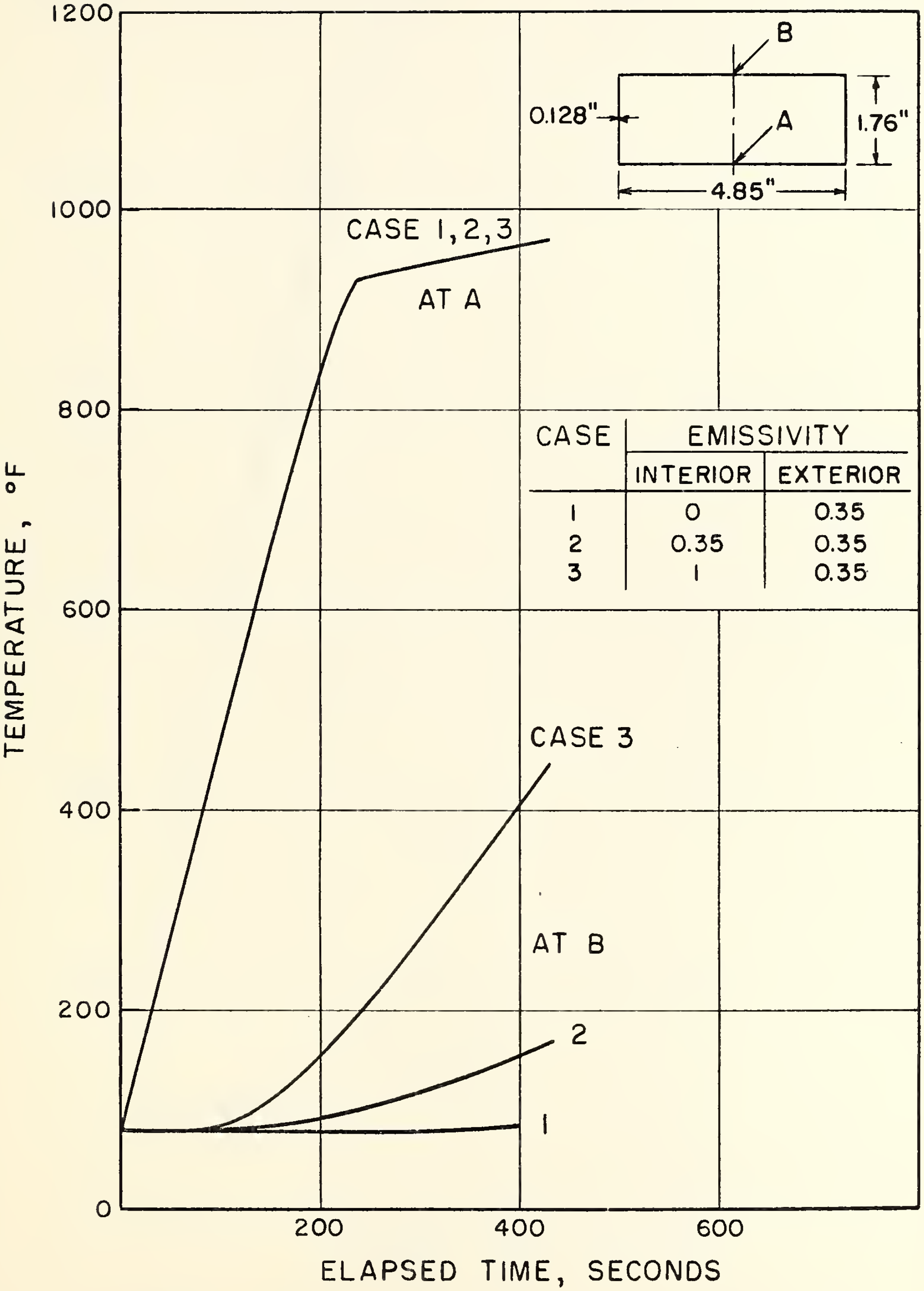
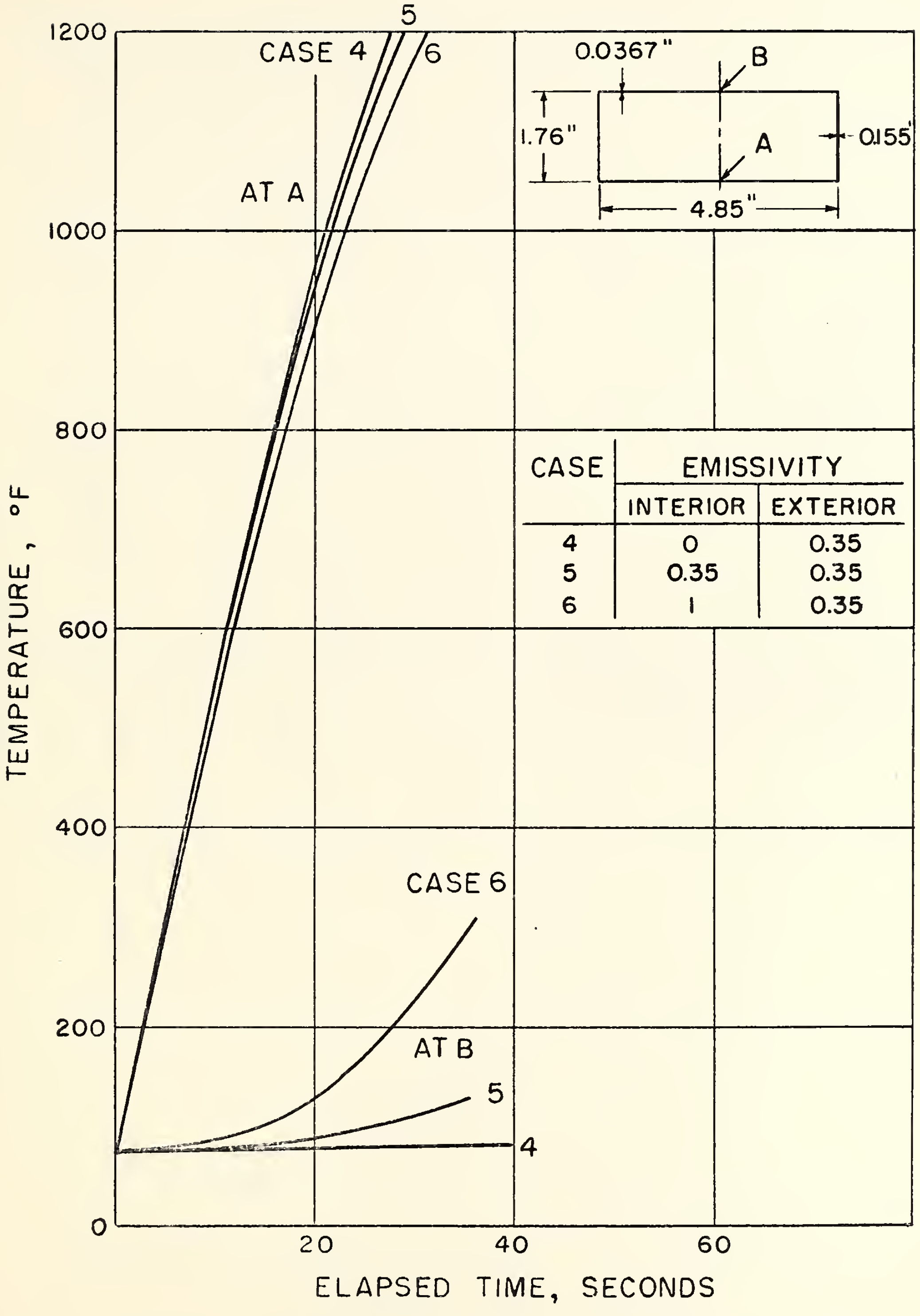


Fig. 1 Test beam cross-section (Cases 1, 2, 3)



CASE	EMISSIVITY	
	INTERIOR	EXTERIOR
1	0	0.35
2	0.35	0.35
3	1	0.35

Fig. 2 BEAM TEMPERATURE HISTORY



CASE	EMISSIVITY	
	INTERIOR	EXTERIOR
4	0	0.35
5	0.35	0.35
6	1	0.35

Fig. 3 BEAM TEMPERATURE HISTORY FOR HEATING RATE OF 32,400 B/HR FT^2

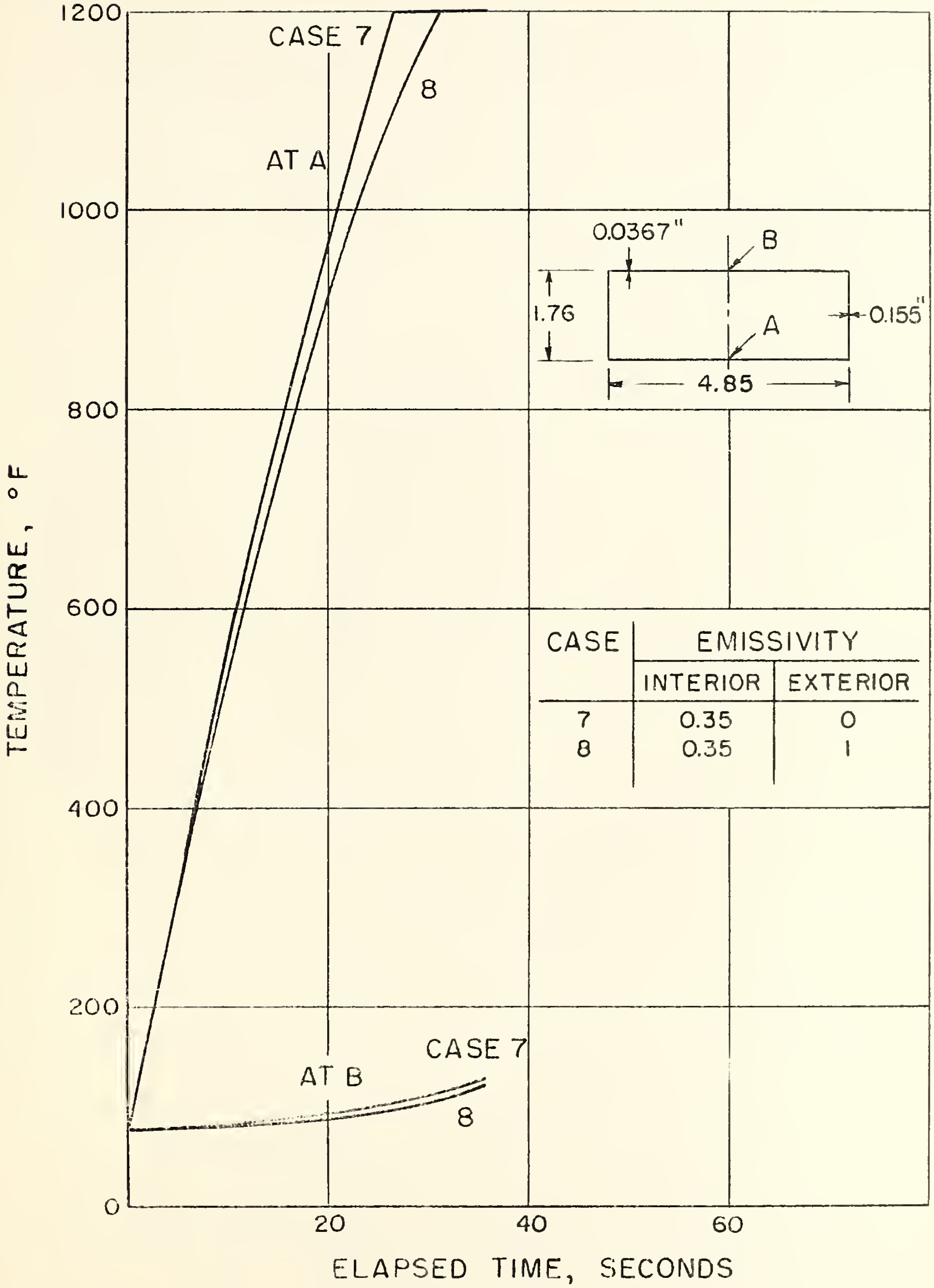
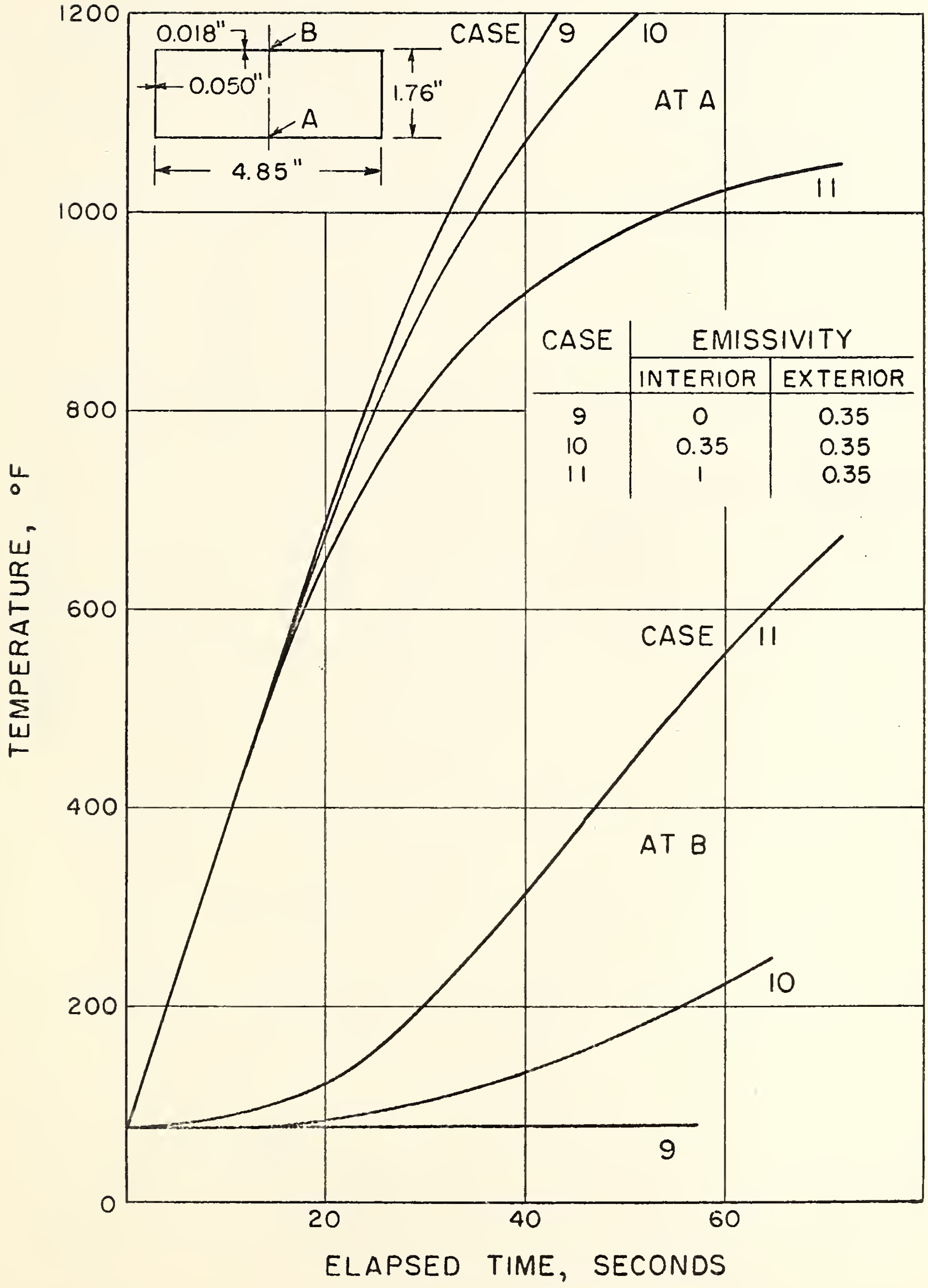
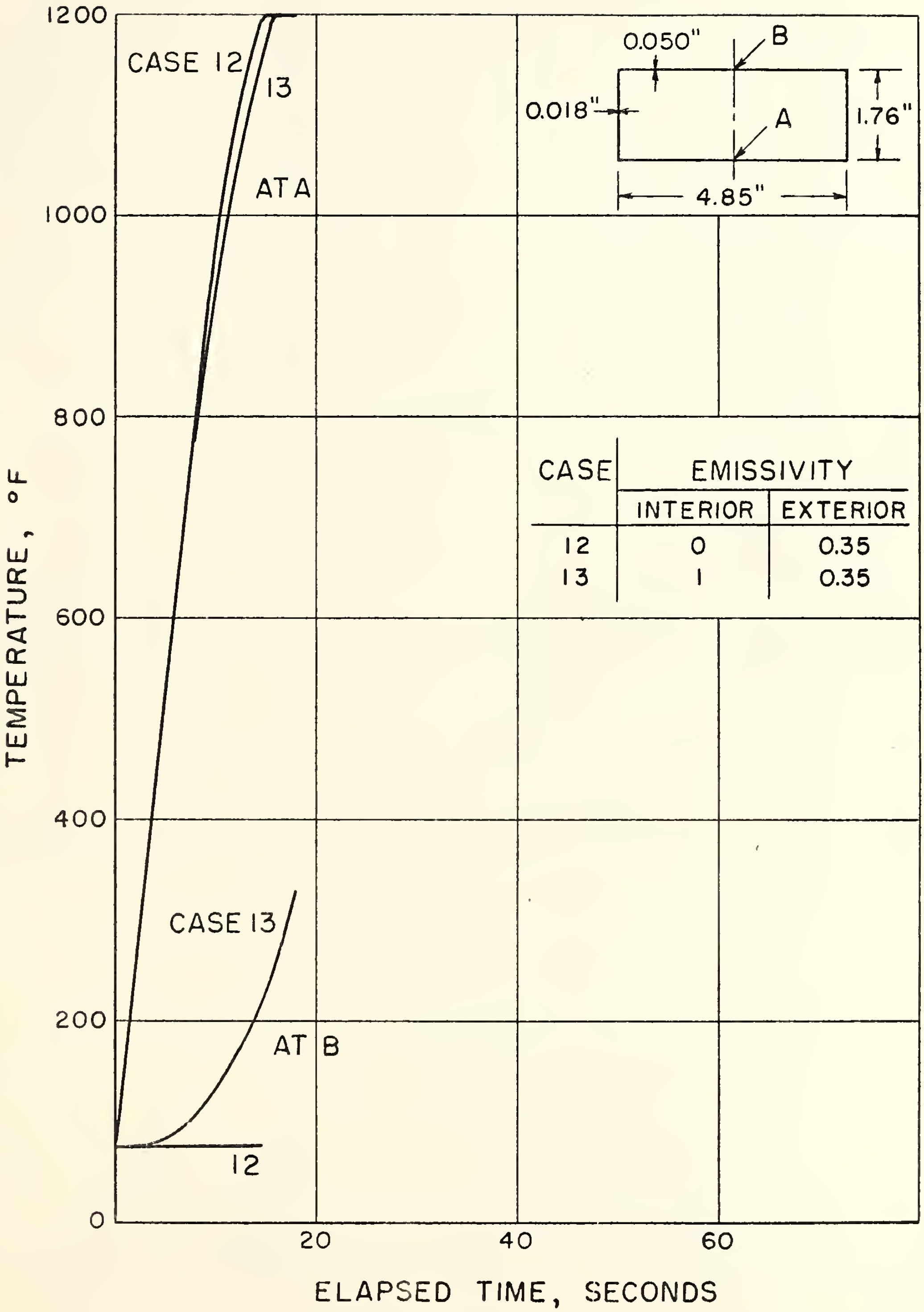


Fig. 4 BEAM TEMPERATURE HISTORY FOR HEATING RATE OF 32,400 B/HR FT^2



CASE	EMISSIVITY	
	INTERIOR	EXTERIOR
9	0	0.35
10	0.35	0.35
11	1	0.35

Fig. 5 BEAM TEMPERATURE HISTORY FOR HEATING RATE OF 10,800 B/HR FT^2



CASE	EMISSIVITY	
	INTERIOR	EXTERIOR
12	0	0.35
13	1	0.35

Fig. 6 BEAM TEMPERATURE HISTORY FOR HEATING RATE OF 32,400 B/HR FT^2

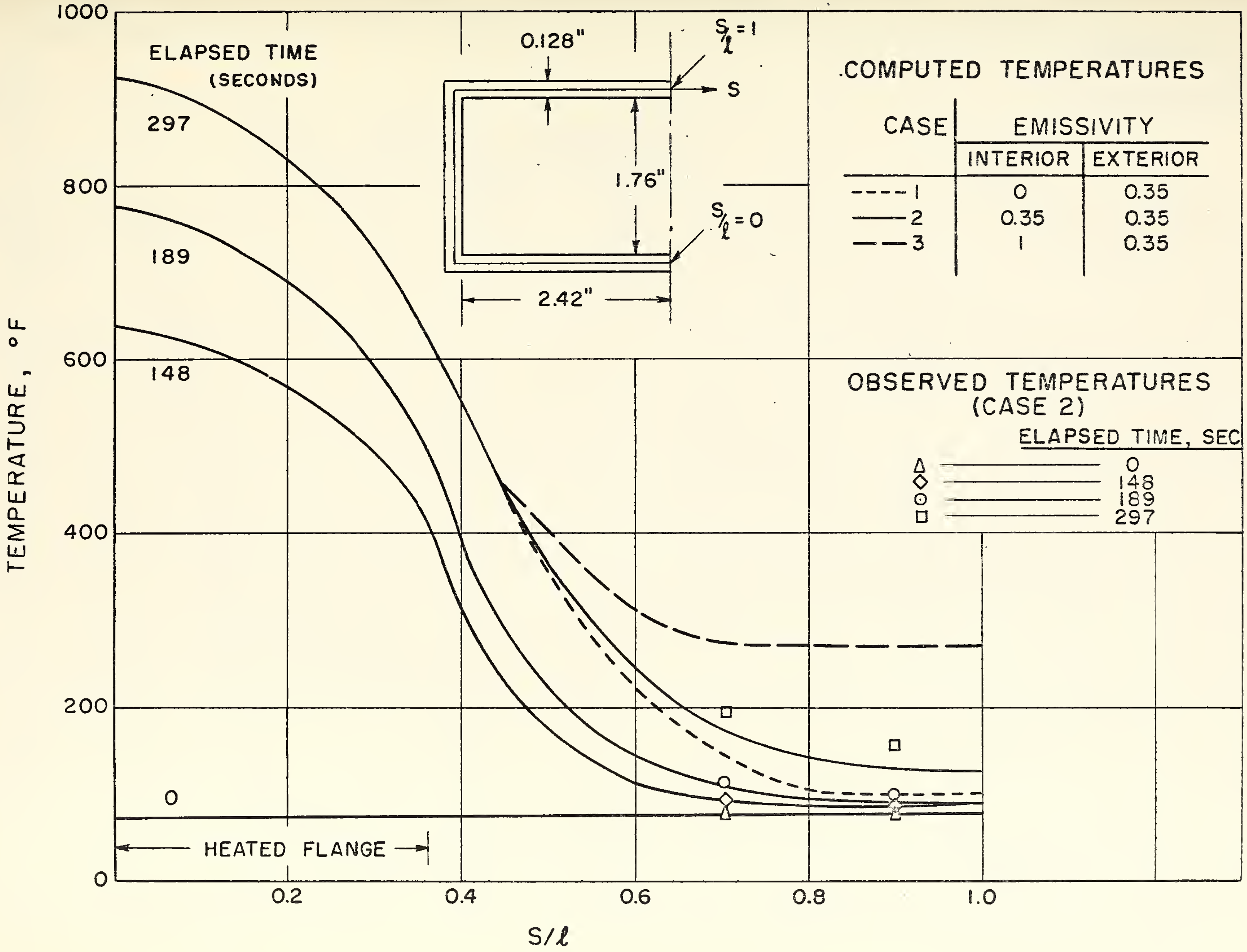
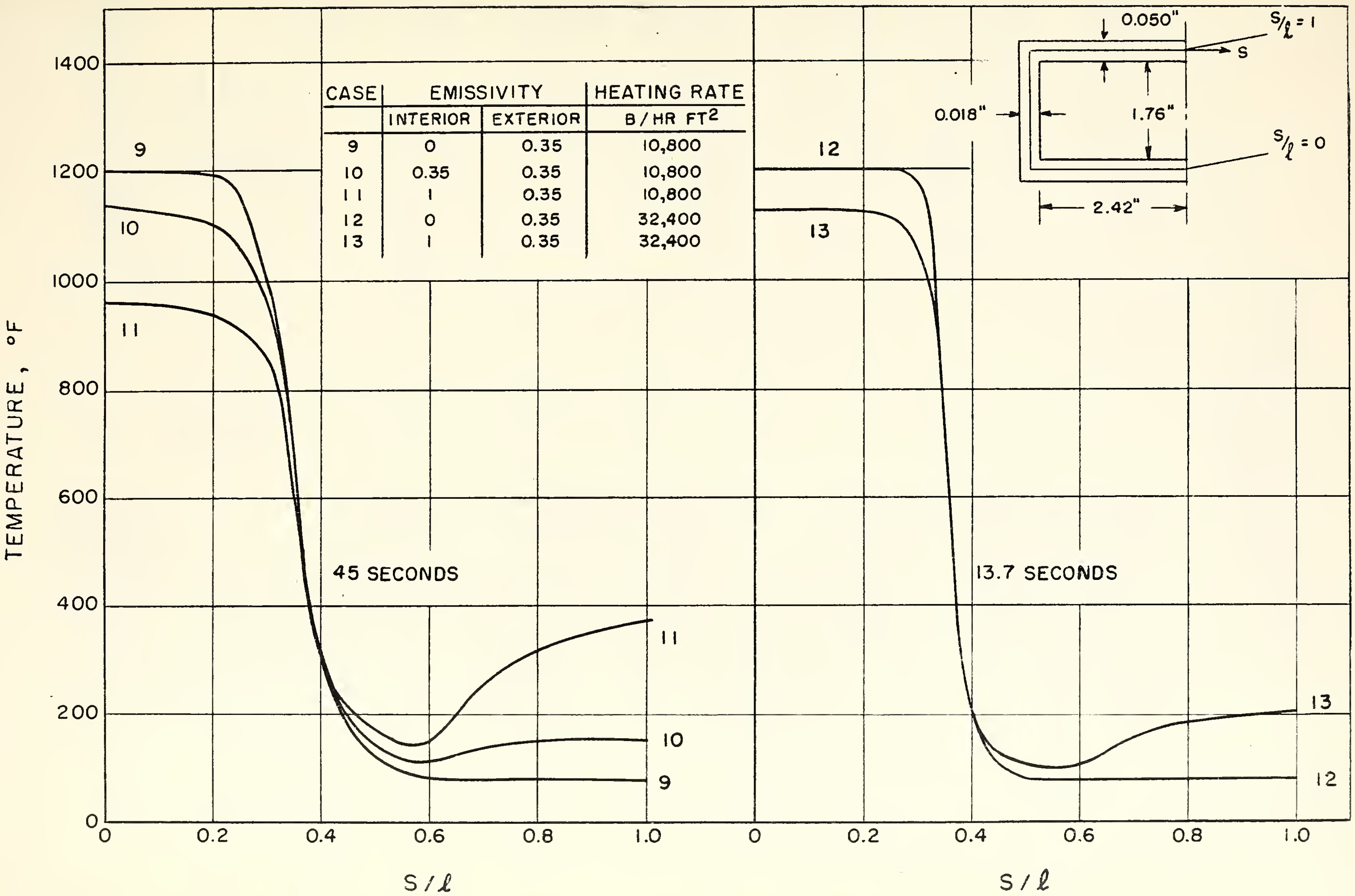


Fig. 7 TEST BEAM TEMPERATURE DISTRIBUTION



CASE	EMISSIVITY		HEATING RATE
	INTERIOR	EXTERIOR	B/HR FT^2
9	0	0.35	10,800
10	0.35	0.35	10,800
11	1	0.35	10,800
12	0	0.35	32,400
13	1	0.35	32,400

Fig. 8 TEMPERATURE DISTRIBUTION IN THIN-WALLED BEAM

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Electricity and Electronics. Resistance and Reactance. Electron Devices. Electrical Instruments. Magnetic Measurements. Dielectrics. Engineering Electronics. Electronic Instrumentation. Electrochemistry.

Optics and Metrology. Photometry and Colorimetry. Optical Instruments. Photographic Technology. Length. Engineering Metrology.

Heat. Temperature Physics. Thermodynamics. Cryogenic Physics. Rheology. Engine Fuels. Free Radicals Research.

Atomic and Radiation Physics. Spectroscopy. Radiometry. Mass Spectrometry. Solid State Physics. Electron Physics. Atomic Physics. Neutron Physics. Nuclear Physics. Radioactivity. X-rays. Betatron. Nucleonic Instrumentation. Radiological Equipment.

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Mechanics. Sound. Mechanical Instruments. Fluid Mechanics. Engineering Mechanics. Mass and Scale. Capacity, Density, and Fluid Meters. Combustion Controls.

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Mineral Products. Engineering Ceramics. Glass. Refractories. Enameled Metals. Concreting Materials. Constitution and Microstructure.

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- Office of Basic Instrumentation.
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